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Vibration Analysis Applied to the Motions of a Gas Turbine Engine

by T. A. Korjack

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Vibration Analysis Applied to the Motions of a Gas Turbine Engine

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Abstract

A method has been suggested to compare and evaluate vibration environments or sinusoidal and random specifications through the use of a single analysis technique. The comparison methodology utilized in this analysis could be significantly implemented and exploited in interpreting spectrum analysis results of vibration data such as in the analysis of a gas turbine engine of a modern tank. A generalized, multi-degree-of-freedom (DOF) lumped parameter structural system model was used to allow an initial evaluation of the environments such as shock, sine, and random so as to possibly eliminate a detailed separate dynamic analysis for each environment. Results indicate a possible reduction of analysis time and costs. In addition, since both sine and random analyses depend heavily on modal damping assumptions for the accuracy of their predictions, the simple methodology proposed herein should prove to be useful and productive.

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1. Introduction

Vibration analysis involving designs should be evaluated for a variety of loads in order to determine the severity and possible failure. Not one environment (viz, shock, sinusoids, or random excitation) allows the design of all structural components and, hence, we are led into separate analyses for shock—random or periodic motions.

This work purports an approach, although quite simple, to compare and evaluate vibration environment or sinusoidal and random specification via a single analysis technique. This comparison methodology will prove to be extremely useful in interpreting spectrum analysis results of vibration data that will be taken of the gas turbine engine of a typical modern tank, especially pertaining to the turbine engine diagnostics (TED) project (Helfman, Dumer, and Hanratty 1995).

2. Methodology

In general, consider an η (multi) degree-of-freedom (DOF) lumped parameter structural system with mass matrix $[M_i]$, stiffness matrix $[k_{ij}]$, damping matrix $[C_{ij}]$, and column matrix of external forces $\{F\ (x_j,\ t)\}$. M_i, k_{ij}, C_{ij}, and F_j are expressed in the w coordinate system. Given a forcing function in the form as

$$\{F\ (x_j,\ t)\} = \{P_o\ p\ (x_j)\}\, f\ (t), \tag{1}$$

then, the differential equations of motion in the w coordinate system take the matrix form

$$[m]\{\ddot{w}\} + [c]\, w + [k]\, w = P_o\ \{p\ (x_j)\}\, f\ (t). \tag{2}$$

If we apply a coordinate transformation,

$$\{w\} = [\phi]\,\{\eta\}, \tag{3}$$

in which each column of ϕ is a modal column of the system and $\{\eta\}$ represents the normal coordinates, then, from equation (3),

$$\{\dot{w}\} = [\phi]\,\{\dot{\eta}\} \tag{4}$$

and

$$\{\ddot{w}\} = [\phi]\,\{\ddot{\eta}\}. \tag{5}$$

Substituting equations (3)–(5) into equation (2) and premultiplying by the transpose of $[\phi]$, equation (2) becomes

$$[\phi]^T\,[m]\,[\phi]\,\{\ddot{\eta}\} + [\phi]^T\,[c]\,[\phi]\,\{\dot{\eta}\} + [\phi]^T\,[k]\,[\phi]\,\{\eta\} = P_o\,[\phi]^T\,\{p\,(x_j)\}\,f(t). \tag{6}$$

From the orthogonality relations of natural modes, it follows that

$$[\phi]^T\,[m]\,[\phi] = [M_r], \tag{7}$$

and

$$[\phi]^T\,[k]\,[\phi] = [\omega_r^2]\,[M_r] = [K_r]. \tag{8}$$

Comparing the triple matrix product

$$[\phi]^T\,[c]\,[\phi] \tag{9}$$

with equations (7) and (8), this product will result in a diagonal matrix only when the damping matrix [c] is proportional to either the mass matrix [m] or the stiffness matrix [k]; that is,

$$[c] = 2\beta\,[M], \tag{10}$$

$$[c] = \alpha\,[k], \tag{11}$$

or

$$\beta r = \zeta_r\,\omega_r, \tag{12}$$

where ζ_r is the ratio of the assumed to critical damping and ω_r is the angular frequency. Using equation (10) in equation (9) and comparing with equation (7),

$$[\phi]^T\ [c]\ [\phi] = 2\beta\,[M_r]. \tag{13}$$

The substitution of equations (7) and (8) with equation (6) results in a set of n decoupled differential equations of motion.

$$[M_r]\{\ddot{\eta}\} + 2\beta\,[M_r]\{\dot{\eta}\} + [\omega_r^2][M_r]\{\eta\} = [\phi]^T\{p\,(x_j)\}\,P_o\,f(t). \tag{14}$$

The rth equation of equation (14) has the form

$$M_r\,\eta_r + 2\beta\,M_r\,\eta_r + \omega_r^2\,M_r\,\eta_r = P_o\,\{\phi(r)\}\,\{p\,(x_j)\}\,f(t). \tag{15}$$

Let

$$\{\phi(r)\}\,\{p\,(x_j)\} = \Gamma_r = \text{participation factor}. \tag{16}$$

Dividing through by M_r in equation (15) and using equation (16) results in

$$\ddot{\eta}_r + 2\beta\,\dot{\eta}_r + \omega_r^2\,\eta_r = \frac{P_o}{M_r}\,\Gamma_r\, f(t). \tag{17}$$

A general form of solution to equation (17) is obtained readily by the use of Laplace transform (Hurty and Rubenstein 1964; Gardner and Barnes 1942). For the case of shock spectrum, the solution to equation (17) becomes

$$\eta_r(t) = \frac{P_o\,\Gamma_r}{\omega_r^2 M_r}\, D_r(t), \tag{18}$$

in which

$$D_r(t) = \int_o^t h(t - \tau)\, f(\tau)\, d\tau$$

$$= \int_o^t \frac{\omega_r^2\, e^{-\beta(t-\tau)}}{\sqrt{\omega_r^2 - \beta^2}}$$

$$= \sin\left[\sqrt{\omega_r^2 - \beta^2}\,(t - \tau)\right] f(\tau)\, d\tau\,, \tag{19}$$

where $D_r(t)$ is called the dynamic load factor.

The total deflection of the structure, w (x,t), is obtained by inserting equation (19) into equation (3). Thus,

$$w(x,t) = P_o \sum_{i=1}^{n} \frac{\Gamma_i}{\omega_i^2\, M_i}\, \phi_i\, D_i(t). \tag{20}$$

Starting again from equation (17), and posing the problem of harmonic or sine excitation,

$$f(t) = e^{i\Omega t} \tag{21}$$

Use of Laplace transform provides the steady-state solution given by

$$\eta_r(t) = \frac{P_o \Gamma_r}{\omega_r^2 M_r} \frac{1}{1 - \frac{\Omega}{\omega_r^2} + i\,2\,\zeta_r \frac{\Omega}{\omega_r}} f(t) \tag{22}$$

$$= \frac{P_o \Gamma_r}{\omega_r^2 M_r} H_r(\Omega)\, f(t), \tag{23}$$

where

$$r = 1, 2, ..., n;\; H_r(\Omega) = \frac{1}{1 - \left(\frac{\Omega}{\omega_r}\right)^2 + i\, 2\zeta_r \frac{\Omega}{\omega_r}}. \tag{24}$$

The total deflection of the structure, w (x;t), is obtained by inserting equation (22) into equation (3). Thus,

$$w(x,t) = P_o \sum_{i=1}^{n} \frac{\Gamma_i}{\omega_i^2 M_i} \phi_i\, H_i(\Omega)\, f(t), \tag{25}$$

or

$$w(x,t) = P_o \sum_{r=1}^{n} \phi_r\, \eta_r(t). \tag{26}$$

The mean square of the response can be written as

$$\overline{w^2(x,t)} = \lim_{T\to\infty} \frac{1}{2T} \int_{-T}^{T} w^2(x,t)\, dt. \tag{27}$$

Substituting in equation (27) from equation (26) and then from equation (23), we have

$$\overline{w^2(x,t)} = \lim_{T\to\infty} \sum_{r=1}^{n} \sum_{s=1}^{n} \phi_r(x)\, \phi_s(x) \frac{1}{2T} \int_{-T}^{T} \eta_r(t)\, \eta_s(t)\, dt$$

$$= \sum_{r=1}^{n} \sum_{s=1}^{n} \frac{P_o^2 \Gamma_r \Gamma_s}{\omega_r^2 \omega_s^2 M_r M_s} \lim_{T\to\infty} \frac{1}{2T} \int_{-T}^{T} H_r(\Omega)\, H_s(\Omega)\, f^2(t)\, dt. \tag{28}$$

If, as an approximation, we disregard phase relations that will tend to result in a higher mean square value in equation (28) (i.e., neglecting terms that are a significant departure from the mean [anomalies]), then the integral on the right-hand side of this equation can be written as

$$\lim_{T\to\infty} \frac{1}{2T} \int_{-T}^{T} H_r(\Omega)\, H_s(\Omega)\, f^2(t)\, dt. \tag{29}$$

When the forcing function f(t) is a representative record of an ergodic process, we can transform the limiting process of equation (29) from the time domain to the frequency domain because the function f(t) is then represented by frequency components in a continuous frequency spectrum $0<\Omega<\infty$. Thus,

$$\overline{f^2(t)} = \lim_{T\to\infty} \frac{1}{2T} \int_{-T}^{T} f^2(t)\, dt = \frac{1}{2\pi} \int_0^{\infty} f(\Omega)\, d\Omega. \tag{30}$$

Using this transformation from the time to the frequency domain in equation (29), we write

$$\lim_{T\to\infty} \frac{1}{2T} \int_{-T}^{T} \left|H_r(\Omega)\right| \left|H_s(\Omega)\right| f^2(t)\, dt = \frac{1}{2\pi} \int_0^{\infty} \left|H_r(\Omega)\right| \left|H_s(\Omega)\right| f(\Omega)\, d\Omega. \tag{31}$$

Substituting from equation (28), we write for the mean square response of an n DOF system excited by a random forcing function:

$$\overline{w^2(x,t)} = \sum_{r=1} \sum_{s=1} \phi_r \phi_s \frac{P_o^2 \Gamma_r \Gamma_s}{\omega_r^2 \omega_s^2 M_r M_s} \times \frac{1}{2\pi} \int_0^{\infty} \left|H_r(\Omega)\right| \left|H_s(\Omega)\right| f(\Omega)\, d\Omega. \tag{32}$$

The solution of equation (32) can be simplified further assume a lightly damped, multi-DOF system, i.e.,

$$H_r(\Omega) = \left[\left(1 - \frac{\Omega}{\omega_r}\right)^2 + 4\zeta_r^2 \frac{\Omega^2}{\omega_r^2}\right]^{-1/2}. \tag{33}$$

According to Hurty and Rebinstein (1964), the magnification factors $\left|H_r(\Omega)\right|$ have regions of pronounced peaks in the neighborhood of the corresponding natural frequencies w_r. Furthermore, Biggs (1964) tells us that the products $\left|H_r(\Omega)\right|$ and $\left|H_s(\Omega)\right|$ for $r \neq s$ are seen to be small in comparison with the same products for r = s. In addition, in equation (32), terms with $r \neq s$ may be negative as well as positive, depending on the sign of the product $\phi_r\ \phi_s\ \Gamma_r\ \Gamma_s$, while terms with r = s are always positive. Equation (32) will therefore be small. Hence, by disregarding the cross-product terms corresponding to r = s, equation (32) becomes

$$\overline{w^2(x,t)} = \sum_{r=1}^{n} \phi_r^2 \frac{P_o^2\,\Gamma_r^2}{\omega_r^4\,M_r^2} \frac{1}{2\pi} \int_o^\infty \left|H_r(\Omega)\right|^2 f(\Omega)\, d\Omega. \tag{34}$$

Equation (34) can be further simplified by approximating the integrals of equation (34) by replacing $f(\Omega)$ by its discrete values $f(\omega_r)$ at the natural frequencies ω_r and using

$$\frac{1}{2\pi} \int_o^\infty \frac{f(\omega_r) d}{\left[1 - \left(\frac{\Omega}{\omega}\right)\right]^2 + 4\zeta^2 \frac{\Omega^2}{\omega^2}} = \frac{f(\omega_r)}{2\pi} \frac{\pi}{4} \frac{\omega_r}{\zeta_r} = \frac{f(\omega_r)\ \omega_r}{8\zeta}. \tag{35}$$

By introducing the following definitions

$$f(\omega_r) = g'_{input} \text{ at } f_n, \tag{36a}$$

$$g'_{in} = g'_{input} = g^2/Hz_{input} \text{ at } f_n, \tag{36b}$$

$$f_n = f_r = 2\pi\omega_r, \tag{37}$$

and

$$Q = \frac{1}{2\zeta}, \tag{38}$$

equation (34) becomes

$$\overline{\ddot{w}^2(x,t)} = \sum_{r=1}^{n} \phi_r^2(x) \frac{P_o^2 \Gamma_r^2}{\omega_r^4 M_r^2} \left(\frac{\pi}{2} g'_{in} f_n Q \right). \tag{39}$$

Equation (39) is the equation that forms the basis of the method of response spectrum analysis. Hence, in this perspective, we have restricted our analysis to the following constraints:

- Ignored phase relations,
- Ignored magnification products at different frequencies,
- Lightly damped all modes (frequencies), and
- Used discrete values of the input spectrum at the natural frequencies.

These conditions are realistic if we are dealing with a system that is grounded by base mounts, and that has harmonic frequencies. At this point, we can summarize and rewrite equations (20), (25), and (39) as follows:

- Case no. 1—shock (from equation [20]):

$$w(x,t) = \sum_{r=1}^{n} P_o \frac{\Gamma_r}{\omega_r^2 M_r} \phi_r(x) D_r(t),$$

where $r = i = 1, 2, \ldots, n$; or from equation (25),

$$\ddot{w}(x,t) = \sum_{r=1}^{n} \frac{P_o \Gamma_r}{\omega^2 M_r} \phi_r(x) D'_r(t),$$

where

$$D'_r\ (t) = \frac{d^2}{dt^2}\ (D_r(t)).$$

- Case no. 2—sine (from equation [39]):

$$w\ (x,t) = \sum_{r=1}^{n} \frac{P_o\, \Gamma_r}{\omega_r^2\, M_r}\ \phi_r\ (x)\ H_r\ (\Omega)\ f(t),$$

where r = i = 1, 2, ..., n; or

$$\ddot{w}\,(x,t) = \sum_{r=1}^{n} \frac{P_o\, \Gamma_r}{\omega_r^2\, M_r}\ \phi_r\ (x)\, Q\, f'\ (t),$$

where

$$f'\ (t) = \frac{d^2}{dt^2}\ (f(t)),$$

and

$$Q = \frac{1}{2\zeta_r}$$

(from equation [33]), when $\Omega = \omega_r$.

- Case no. 3—random:

$$\overline{\ddot{w}^2\,(x,t)} = \sum_{r=1}^{n} \frac{P_o^2\, \Gamma_r^2}{\omega_r^4\, M_r^2}\ \phi_r^2\ (x) \left(\frac{\pi}{2}\ g'_{in}\ f_n\ Q \right), \tag{40}$$

where, in all cases, $\ddot{w}\,(x,t)$ is defined as response acceleration. In case no. 3, response g_{RMS} is defined as

$$g_{RMS} = \sqrt{\overline{\ddot{w}^2\,(x,t)}} = \left[\sum_{r=1}^{n} \frac{P_o^2\, \Gamma_r^2}{\omega_r^4\, M_r^2}\ \phi_r^2\ (x) \left(\frac{\pi}{2}\ g'_{in}\ f_n\ Q \right) \right]^{1/2}, \tag{41}$$

which includes all frequencies (w_r, f_n) and modes $\phi_r(x)$ of the structural system under consideration. If we look at g_{RMS} per mode or frequency, we are led to

$$g_{RMS_{\omega_r}} = \sqrt{\overline{\ddot{w}^2(x,t)}} = \frac{P_o \Gamma_r}{\omega_r^2 M_r} \phi_r(x) \left(\frac{\pi}{2} g'_{in} f_n Q \right)^{1/2} \tag{42}$$

as a first step toward the overall g_{RMS} computation. By comparing equation (41) to case no. 2 and, in turn, to case no. 1, we see that the only difference in the response of a multi-DOF structure at a natural frequency, ω_r, are the forcing function terms; that is,

case no. 1—shock:

$$D'_r(t), \tag{43}$$

case no. 2—sine:

$$Q f'(t), \tag{44}$$

and case no. 3—random:

$$\left(\frac{\pi}{2} g'_{in} f_r Q \right)^{1/2}. \tag{45}$$

All other terms of the aforementioned cases; that is,

$$\frac{P_o \Gamma_r}{\omega_r^2 M_r} \phi_r(x), \tag{46}$$

define a multi-DOF system (as opposed to a one-DOF system) and are common to all $g_{response}$ solutions for shock, sine, or random environments.

Equations (42)–(44) are nothing more than the $g_{response}$ terms as discussed for a simple one-DOF system by Hurty and Rubinstein (1964).

Using the analysis technique for response spectrum analysis and saving the g_{rms} terms at each natural frequency will then allow the determination of true g_{rms} response for each environment (shock, sine, or random) that differs by a g_{input} magnitude difference defined and known via equations (42)–(44).

3. Conclusions

The approach shown previously provides the benefits of time and cost savings but is limited by the assumption required to derive the approach. The application of the approach can be verified early in the design but requires the capability to solve the dynamic analysis using a program that utilizes both approaches formulated by equation (28) vs. equation (39).

Notwithstanding the fact that shock response is little affected by assumed damping, both sine and random analyses depend heavily on modal damping assumptions for the accuracy of their predictions. Although the approach in this analysis was based upon the concept of response spectrum analysis, its concept is used quite extensively in oscillatory systems and even in earthquake analysis. Here, the simplified equations can be directly correlated to sine or shock vibration analyses. Hence, by means of this simplified, but yet practicable technique, one can easily solve all or any one of the designing environments for a given structure, such as the case for the mechanical structural components/mechanisms inherent in a typical gas turbine engine assembly network.

However, this approach is feasible provided that phase relations are ignored and magnification factors are not a problem. Furthermore, this methodology is especially useful for systems involving harmonic frequencies based upon the natural frequencies of the individual mechanical components.

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4. References

Biggs, J. M. *"Introduction to Structural Dynamics."* McGraw-Hill, 1964.

Gardner, M. F., and J. L. Barnes. *"Transients in Linear Systems."* John Wiley Sons, Inc., New York, NY, 1942.

Helfman, R., J. Dumer, and T. Hanratty. "Turbine Engine Diagnostics." ARL-TR-856, U.S. Army Research Laboratory, Aberdeen Proving Ground, MD, September 1995.

Hurty, W. C., and M. F. Rubinstein. *"Dynamics of Structures."* Prentice-Hall, Inc., Englewood Cliffs, NJ, 1964.

Thomson, W. T. *Laplace Transforms* (2nd ed.). Prentice-Hall, Inc., Englewood Cliffs, NJ, 1960.

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